Computer aided analysis and design of hoisting mehanism of an EOT crane

A THESIS SUBMITTED IN PARTIAL FULFILLMENT OF THE REQUIREMENTS FOR THE DEGREE OF

Bachelor of Technology

(Mechanical Engineering)

by

Pradyumnakesharimaharana

(Roll Number: 108ME022)

DEPARTMENT OF MECHANICAL ENGINEERING

NATIONAL INSTITUTE OF TECHNOLOGY ROURKELA

ROURKELA 769008

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CERTIFICATE

This is to certify that thesis entitled, "***Computer aided analysis and design of hoisting mechanism of an EOT crane***" submitted by Mr. Pradyumna Keshari Maharanain partial fulfillment of the requirements for the award of degree of *Bachelor of Technology* in Mechanical Engineering at National Institute of Technology, Rourkela is an authentic work carried out by him under my supervision and guidance.

To the best of my knowledge, the matter embodied in this thesis has not been submitted to any other university/ institute for award of any Degree or Diploma.

Date: 14 . 05 .2012 Prof. N. Kavi

Professor, Mechanical Engineering Department

Rourkela National Institute of technology

A C K N O W L E D G E M E N T

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Date: 14/05/2012 PradyumnaKeshariMaharana

Place: NIT Rourkela Roll No: 108ME022

Abstract

In this project an overall design of the hoisting mechanism of an EOT crane has been carried out. The dimensions of the main components have been determined for a load capacity of 50 ton crane having 8 rope falls . Various dimensions for cross sections of various shapes for crane hook have been found. After the system was designed ,the stress and deflection are calculated at critical points using ANSYS and optimized. which cross section would be better keeping some parameters constant for all the case . Various dimensions and load per wire for wire ropes has been found. Using various formulae found the dimensions for pulley, Rope-drum . Also calculated the Power and ratings for the motor ,brakes used in the hoist mechanism.

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INTRODUCTION

Hoisting is the process of lifting something or some load or person from lower posion to higher position with the help of some device or mechanism

Hoisting Devices

A hoisting device is used for lifting or lowering a load by means of a drum or lift-wheel around which rope or chain wraps. It may be manually operated, electrically or pneumatically driven and may use chain, fiber or wire rope as its lifting medium

Eg: Elevators ,crane

Here the hoisting part of an EOT crane is discussed

The hoisting part of the EOT crane consists of the following parts

- Hoist motor
- Gear box
- Drum
- Pulleys
- Wire rope
- Hook

A hoist motor is used as a driving system for the mechanism

The motor is coupled to a gearbox

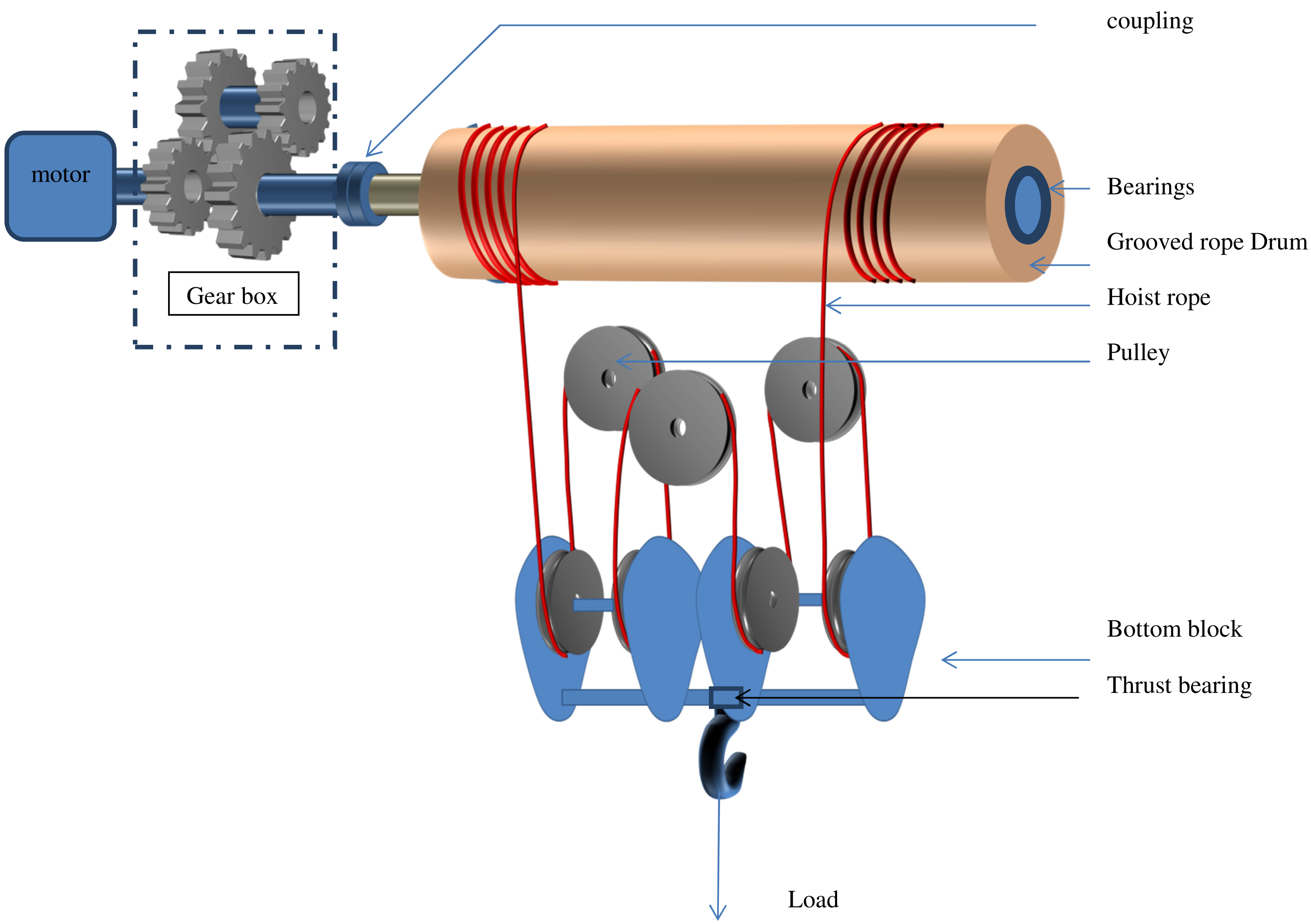


Fig. 1: Schematic view of the hoisting device

The gear box is coupled to the rope drum

The rope is wounded on the rope drum

The pulleys are arranged with 8 rope falls

At the bottom of the pulley the hook is attached with the help of a thrust bearing

Design of the Crane Hook

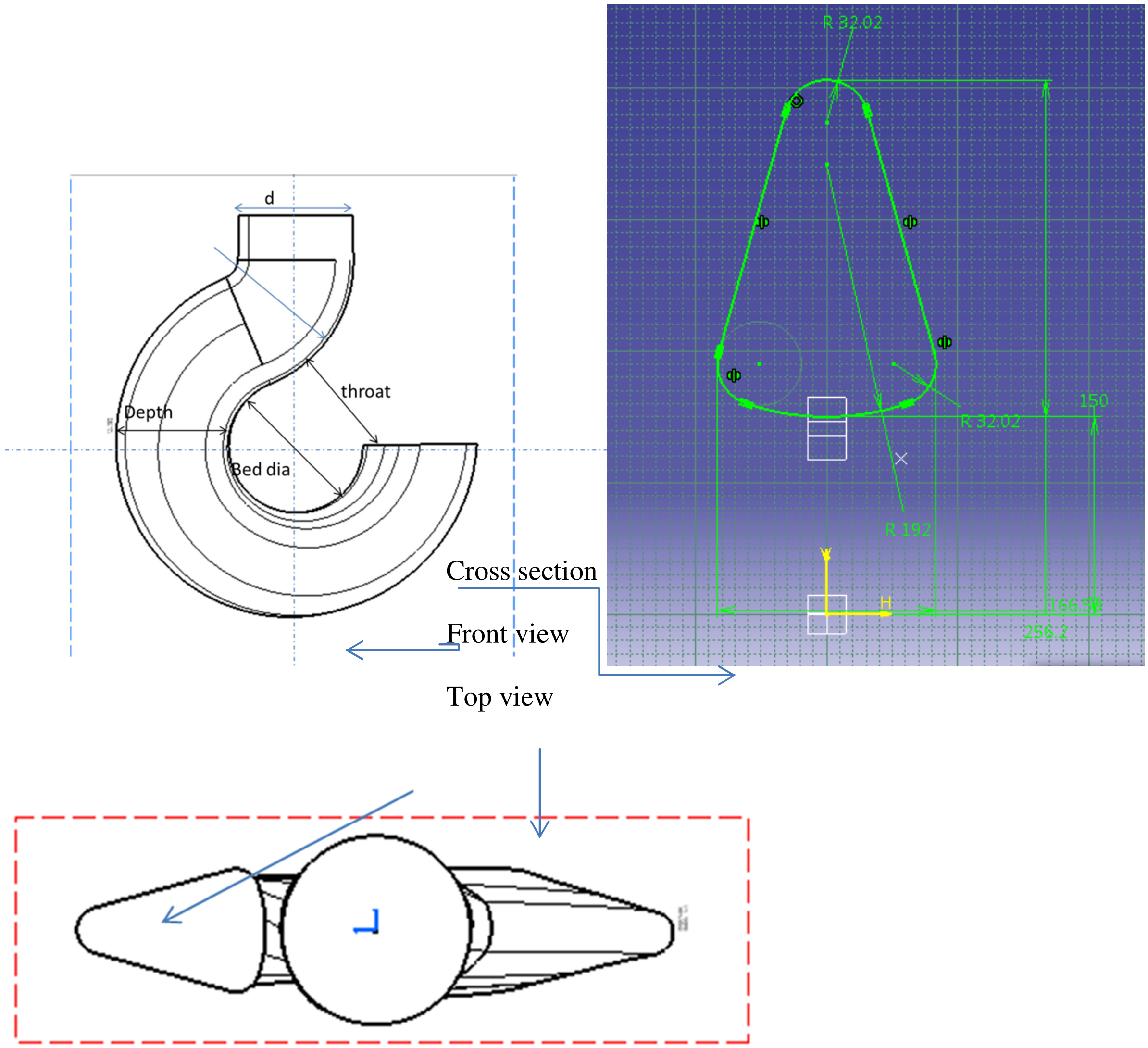


Fig 2 : different views of the crane hook

The inner side is called intrados and the outer side is called the extrado

According to force diagram of the hook intrado experiences more tensile force than the extrado

Hook bed diameter is given by the formula $c = \mu\sqrt{P}$ cm

Where P is the load applied in tonne c is the bed diameter

µis a constant varying from 3.8 to 7.6

Considering µ= 4.24 c = 30cm =300mm

Throat of the hook is taken 0.75c =225mm

d= $3.2\sqrt{P} + c/10$ =256.2mm

Using safety factor 6

W=6x50000 =300,000 N

- A crane hook is treated as a curved beam
- Straight beam theory and shallow beam bending theory is not applied to the crane hook .Bending theory of beam with larger curvature is applied here
-

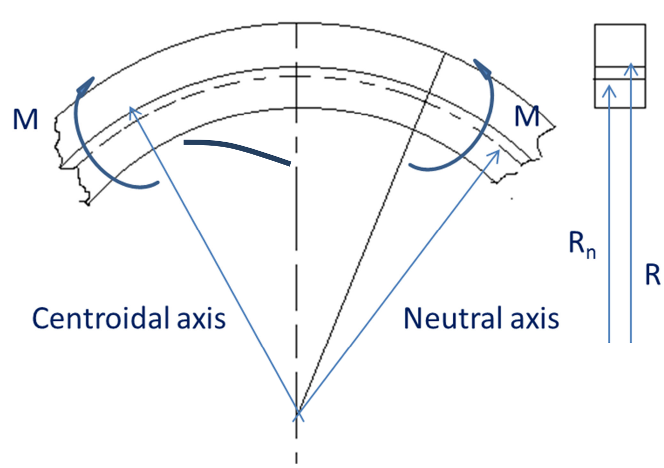


Fig 3: bending of a beam with larger initial curvature

- In a beam with larger curvature neutral surface is displaced from the passing through the centroid towards the center of curvature.

The stress distribution in the curved beam due to moment is found by balancing the internal resisting moment to the externally applied moment

And we get the following result

------------ (Eq^n 1)

$$\sigma = \frac{My}{Ae(R_n - y)}$$

By simplifying putting value of R_n we can obtain the following equation

$$\sigma = \frac{M}{AR}\left[1 - \frac{R^2 y}{h^2(R - y)}\right]$$

------------ (Eq^n 2)

[Where h^2 is called link radius and the value is given by the

formula $h^2 = \frac{R}{A}\int \frac{y^2}{R+y} dA$] ------------ (Eq^n 3)

If σ is +ve then stress is tensile if σ is –ve then the stress is negative

The value of y is taken as –ve if it is on the opposite of the neutral axis

Cross Section	Value of link radius
rectangular	$h^2 = 2.3\frac{R^3}{D}\log\left(\frac{2R+D}{2R-D}\right) - R^2$
circular	$h^2 = \frac{d^2}{16} + \frac{1}{8} \times \frac{d^4}{16R^2}$
trapezoidal	$h^2 = \frac{R^3}{A}\left[2.3\left(B_2\frac{(B_1-B_2)R_2}{D}\right)\log\frac{R_2}{R_1} - (B_1-B_2)\right] - R^2$
triangular	$h^2 = \frac{R^3}{A} \times \frac{B}{D}\left[2.3R_2\log\frac{R_2}{R_1} - D\right] - R^2$

Modified

Section

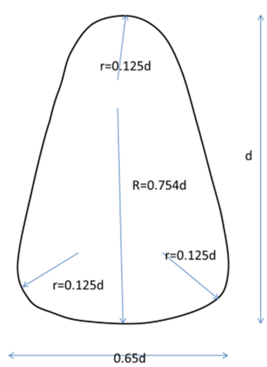


Fig 4.1

d= $3.2\sqrt{P} + c/10$

=256.2mm

R=0.75 d =192.1

r =1/8d =32.05

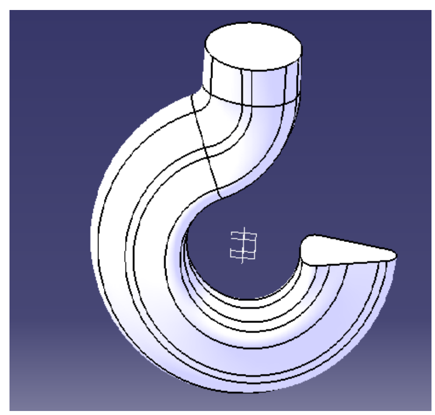


Fig 4.2

Fig 4: Modified cross section

Stress Calculation

Figures

Circular cross section

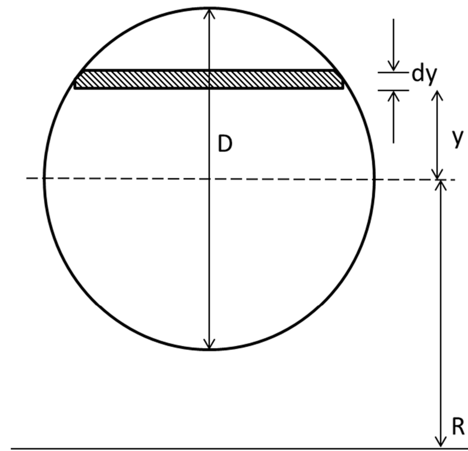


Fig 4.1

D=d=256.2

A=51526.1 mm^2

R=c+d/2= 428 h^2 =4280.89

$$\sigma = \frac{W}{A} + \frac{M}{AR}\left[1 - \frac{R^2 y}{h^2(R-y)}\right]$$

Considering $M \approx W \times R$ we can have

$$\sigma = \frac{W}{A}\left[2 - \frac{R^2 y}{h^2(R-y)}\right]$$

σ = -368.7 for y =d/2

For y= -d/2

Rectangular Cross section

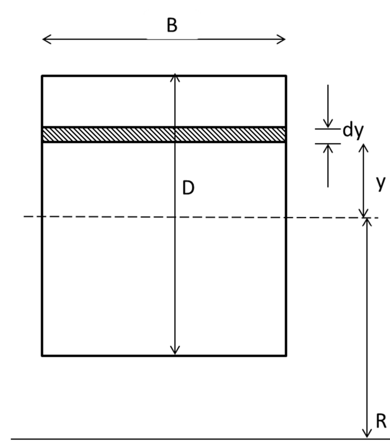


Fig 4.2

D= 256.2 mm

A=51562.1mm^2

B=120.4

$$h^2 = 2.3\frac{R^3}{D}\log\left(\frac{2R+D}{2R-D}\right) - R^2$$

$$h^2 = 5626$$

$\sigma = 12.50$ for y =d/2

Triangular cross section

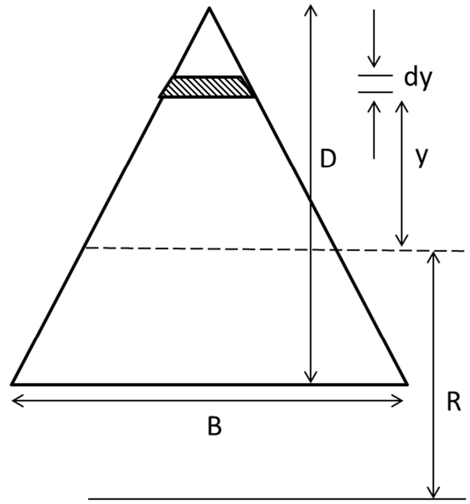


Fig 4.3

D=256.2 mm

B=402.5

A=51552 mm^2

$$h^2 = \frac{R^3}{A} \times \frac{B}{D}\left[2.3R_2 \log\frac{R_2}{R_1} - D\right] - R^2$$

Considering $M \approx W \times R$ we can have

$$\boldsymbol{\sigma = 17.97}$$

Trapezoidal cross section

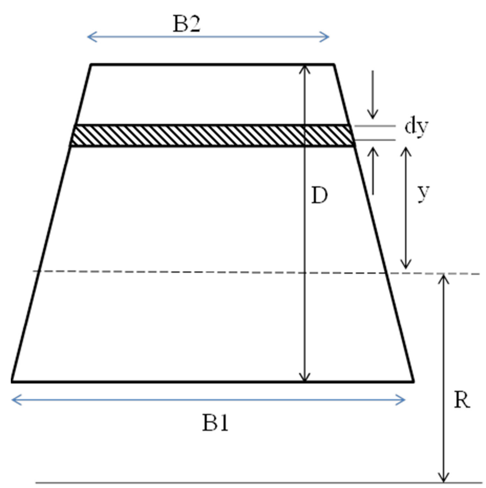


Fig 4.4

B1=300

B2=102.4

$$h^2 = \frac{R^3}{A}\left[2.3\left(B_2 \frac{(B_1 - B_2)R_2}{D}\right)\log\frac{R_2}{R_1} - (B_1 - B_2)\right] - R^2$$

Considering $M \approx W \times R$ we can have

σ =16.35

The below figures show the ANSYS analysis of crane hooks of various cross sections

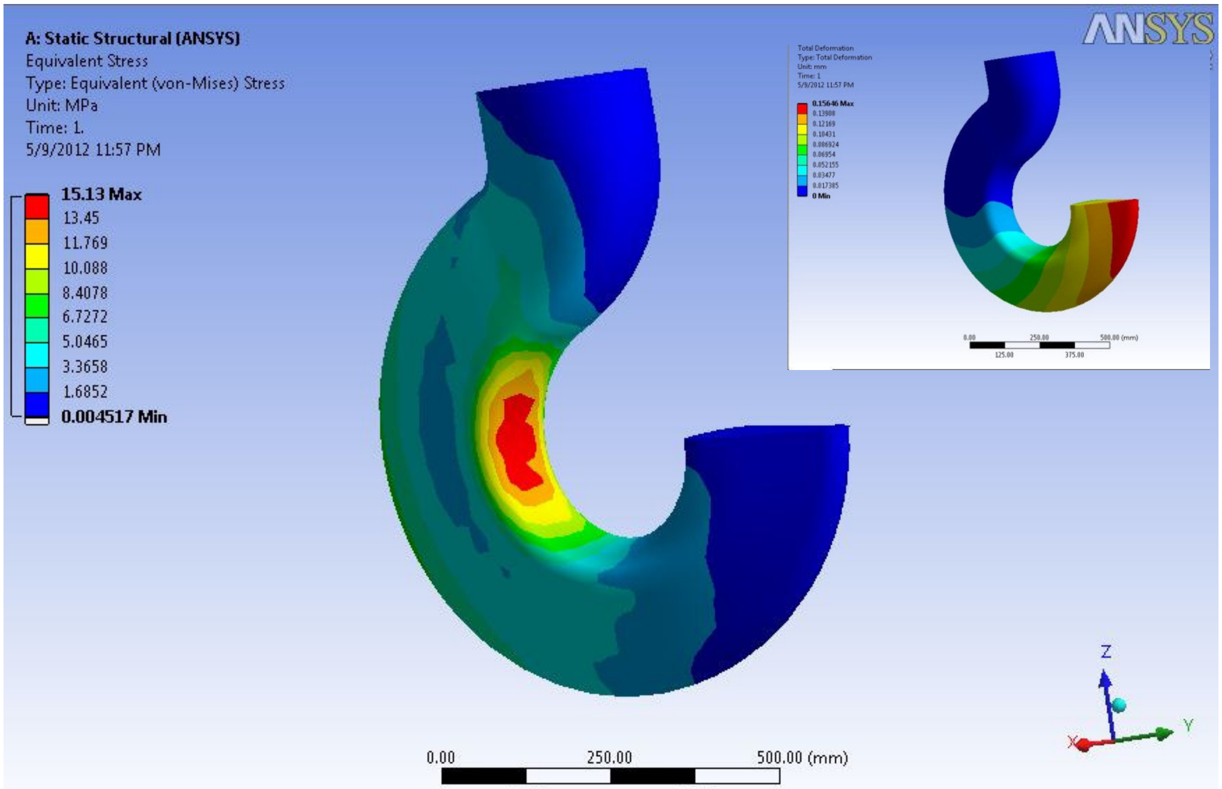


Fig 6 : Stress distribution in the hook with modified cross section

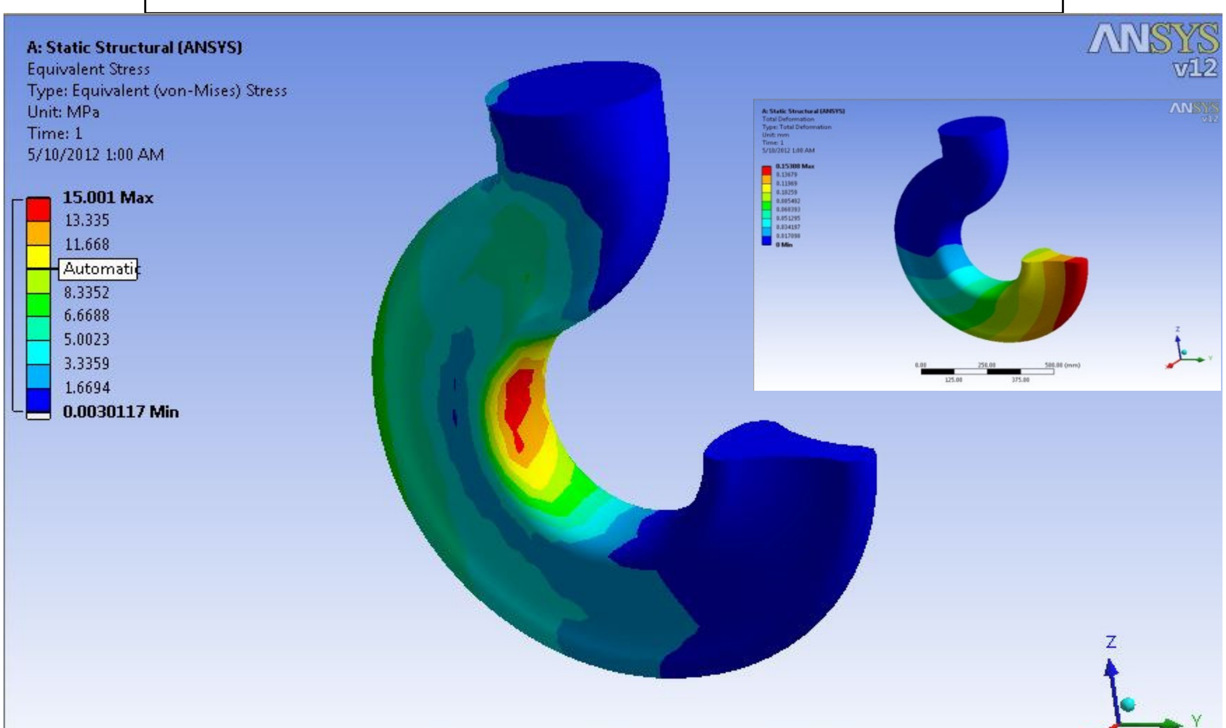


Fig 7 : Stress distribution in the hook with another modified cross section

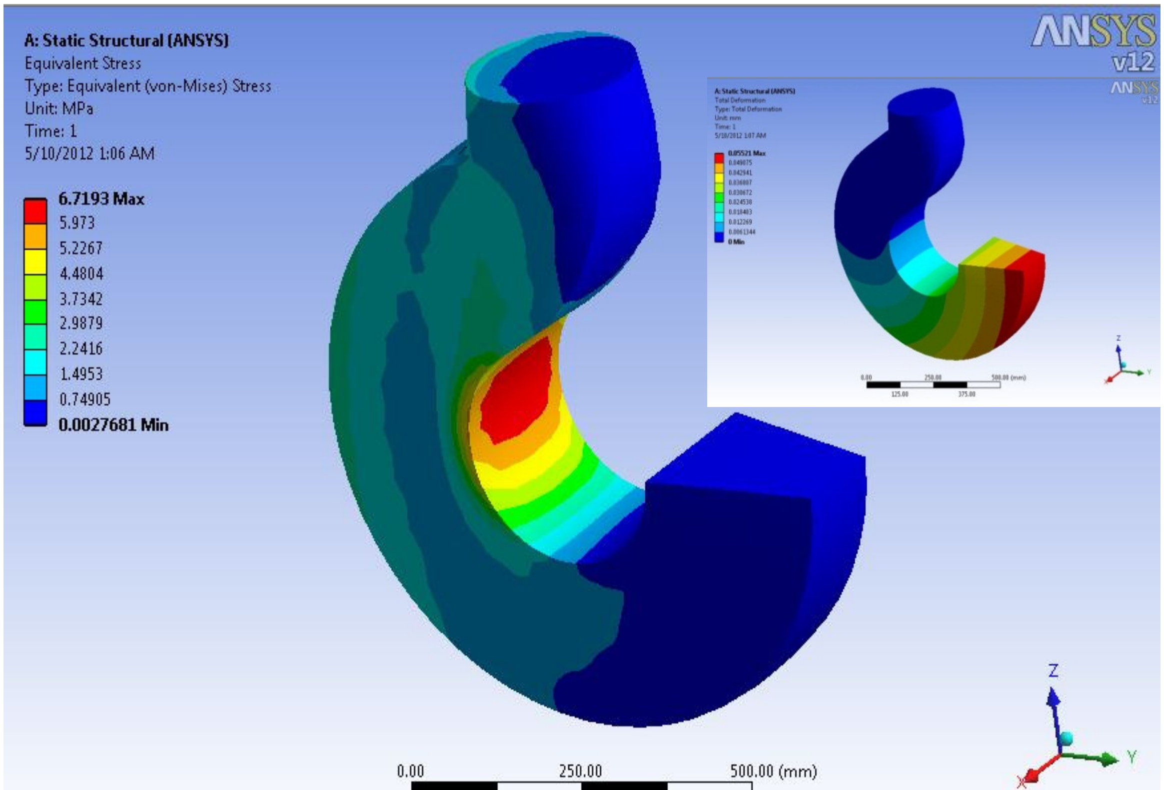


Fig 8 : Stress distribution in the hook with Trapezoidal cross section

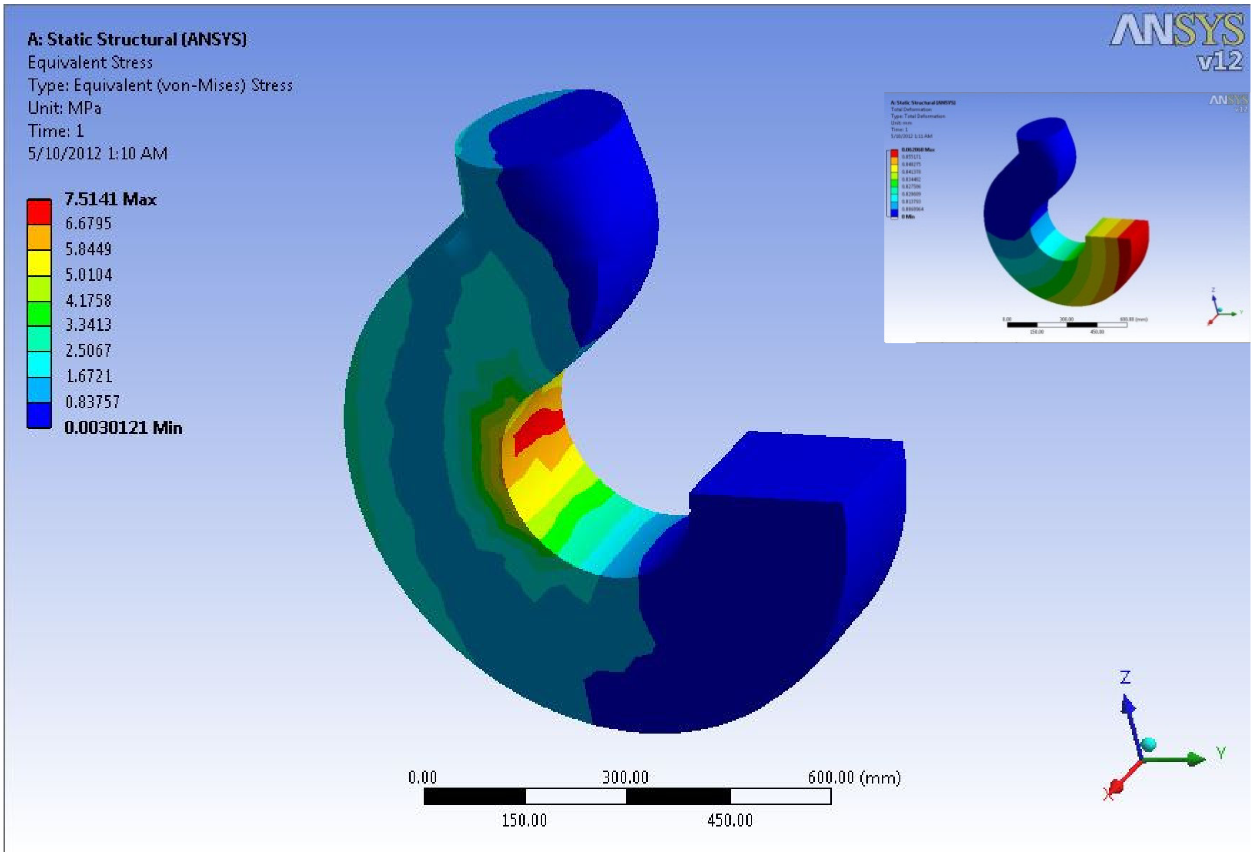


Fig 9 : Stress distribution in the hook with Rectangular cross section

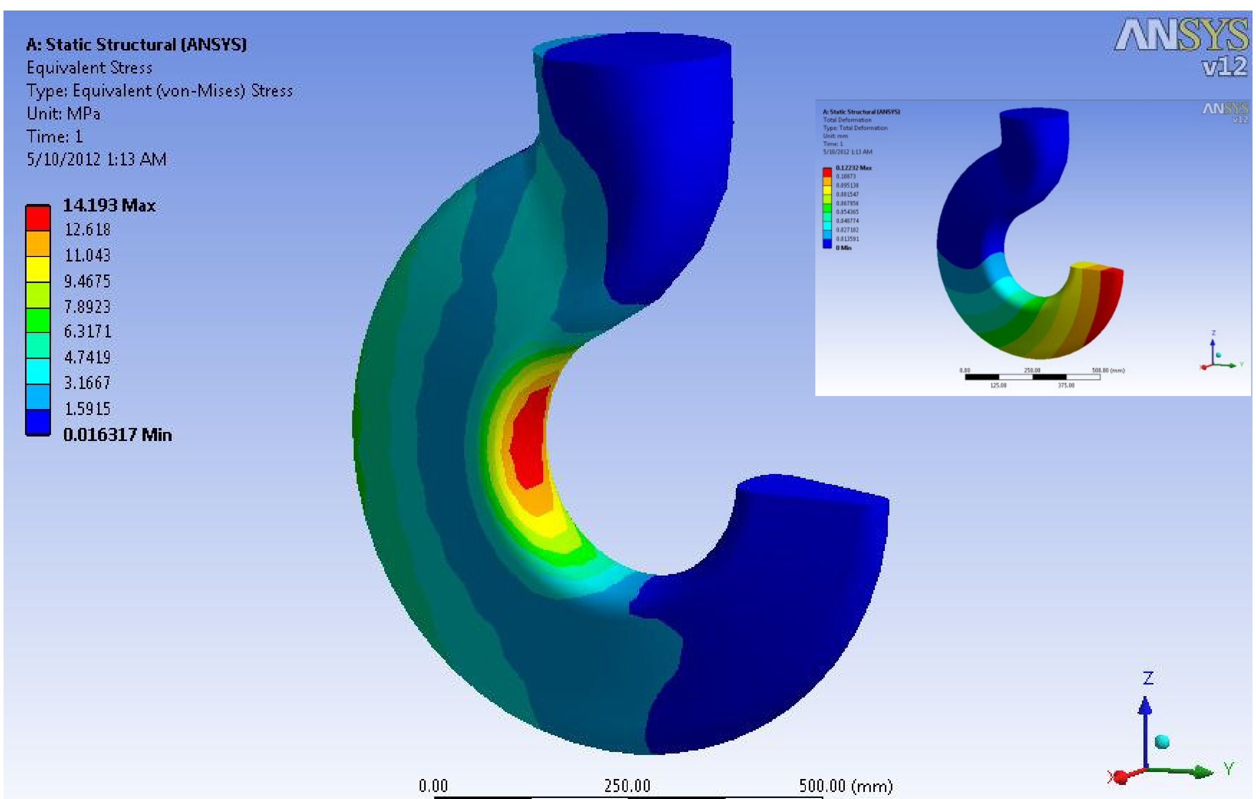


Fig 10 : Stress distribution in the hook with with a modi fied section

Design of the Hoist Wire Rope

A hoisting device use chain, fiber or wire rope as its lifting medium

Wire rope consists of several strands laid (or ‘twisted’) together like a helix.

Each strand is likewise made of metal wires laid together like a helix.Abrasion resistance increases with fewer,larger outside wiresper strand. And fatigue resistance increase with more outside smaller wires per strand

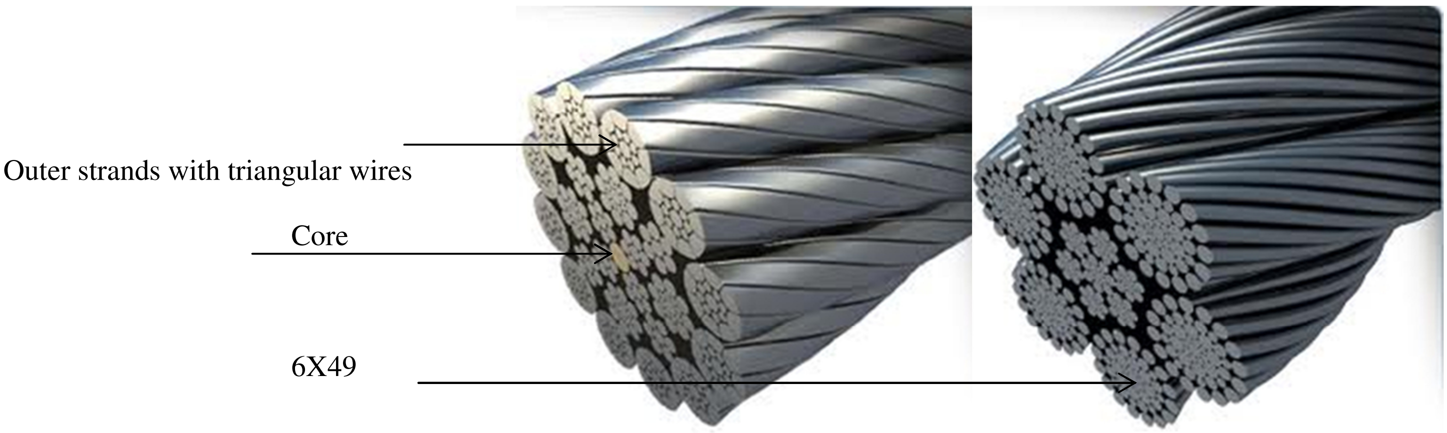
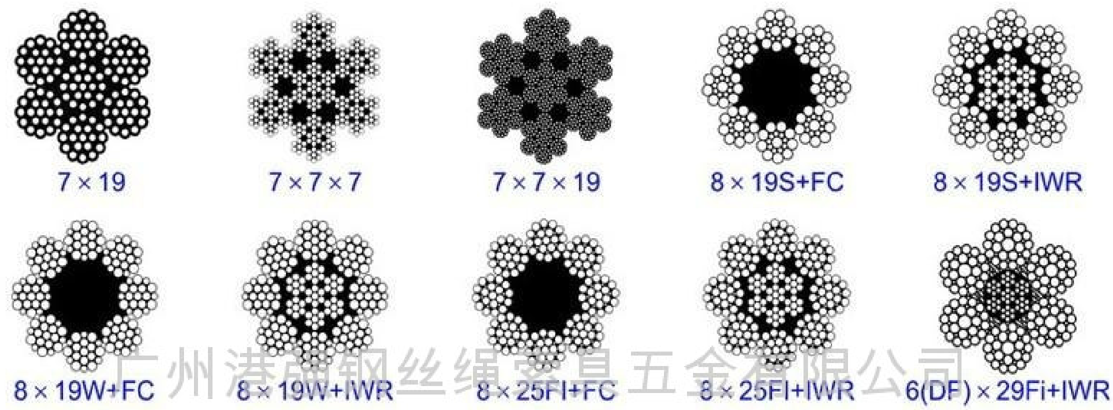


Fig 11:Wire Rope and its strands

Abrasion resistance increases with fewer,larger outside wiresper strand

Abrasion resistance increases with fewer,larger outside wiresper strand. And fatigue resistance increase with more outside smaller wires per strand

Fig 12 : Typical cross sections of a wire Rope



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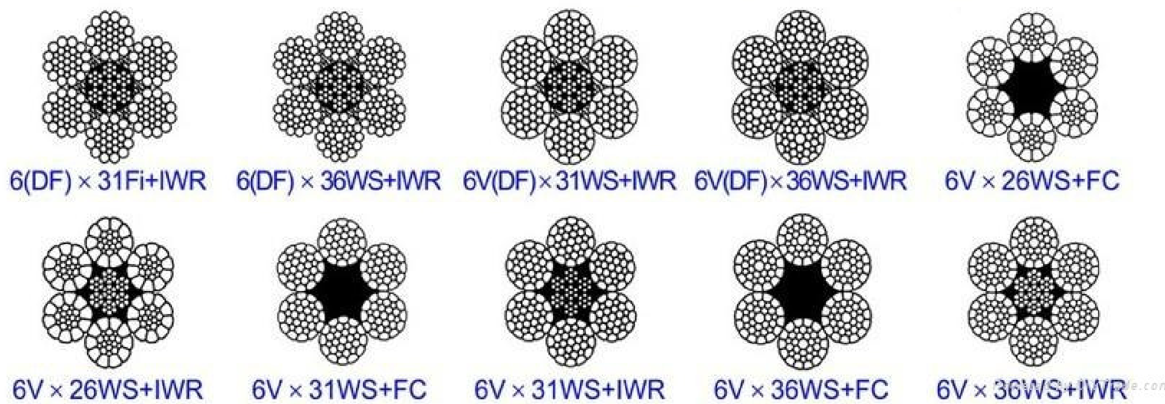


Refered from http://www.diytrade.com/china/pd/9098053/trianfular_strand_wire_rope.html

Ropes are used in various fashion

We can use different no of rope falls

Increase in no of rope fall helps in decreasing the tension in the wire rope

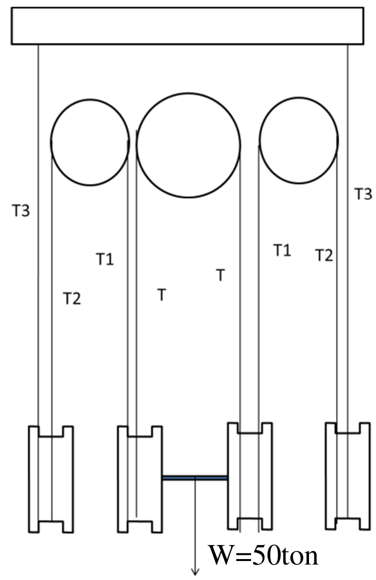
But due to friction if we increase no of rope falls then the tue to friction force tension also increases

Increase in one rope fall increases the tension by c times

C is a coefficient factor which varies from 1.03 to 1.14

Advantage of rope fall is that amount of work done is more incase of more rope falls

Work done = F x S



According to theory with increase in no of rope falls the thension in the rope will be given as follows

$$T1= cT$$
$$T2=cxT1$$
$$T3=CxT2$$

Fig 13:Pulley arrangement using 8 rope falls

$2x(c^3+c^2+c+1)T=50$ ton x F.S.(assume 6) [where c=coefficient of resistance between rope and pulley]

Tension =35.327 ton

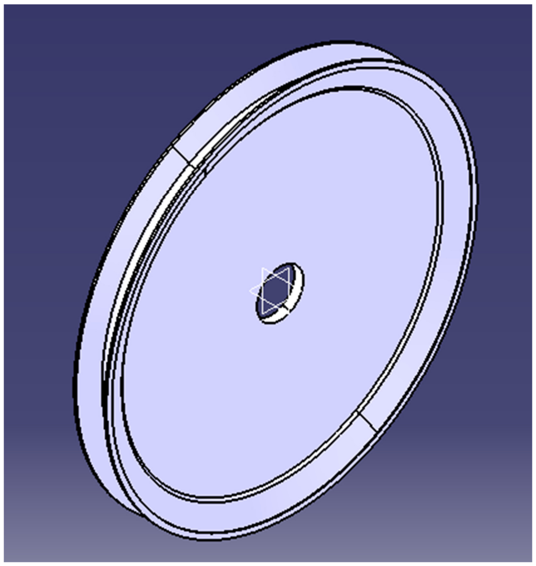
Rope type	Brakeing load	Rope diameter In cm	Available dimension(in mm)	Weight of rope/10 m in kg	Effective area	Stress in a wire
6x7	$4800d^2$	2.71	29		51%	1.68tonne/14.39mm^2 =116.74N/ mm^2
6x19	$5100d^2$	2.63	29	29.210	50%	0.61tonne/6.39mm^2 =95 N/ mm^2
6x37	$4800d^2$	2.71	29	29.332	43%	0.37tonne/2.84 mm^2 =130 N/ mm^2
8x19	$4400d^2$	2.83	29	30.212		

The optimized rope will be 6 x19

Using of wire rope with more no of wire in strands increase the flexibility of the

Pulley

Pulley material : cast iron



Pulley dia = 18 d to 27 d

Taking 25d = 25x29 =725mm

Pulley diameter= 725 mm

symbol	Value in tems of d	value
d		29mm
t	0.4d	11.6mm
h	1.6d	46.4mm
B1	2.7d	70.3
B2	2.1d	60.9
l	0.75d	19.7
R0	1.1d	31.9

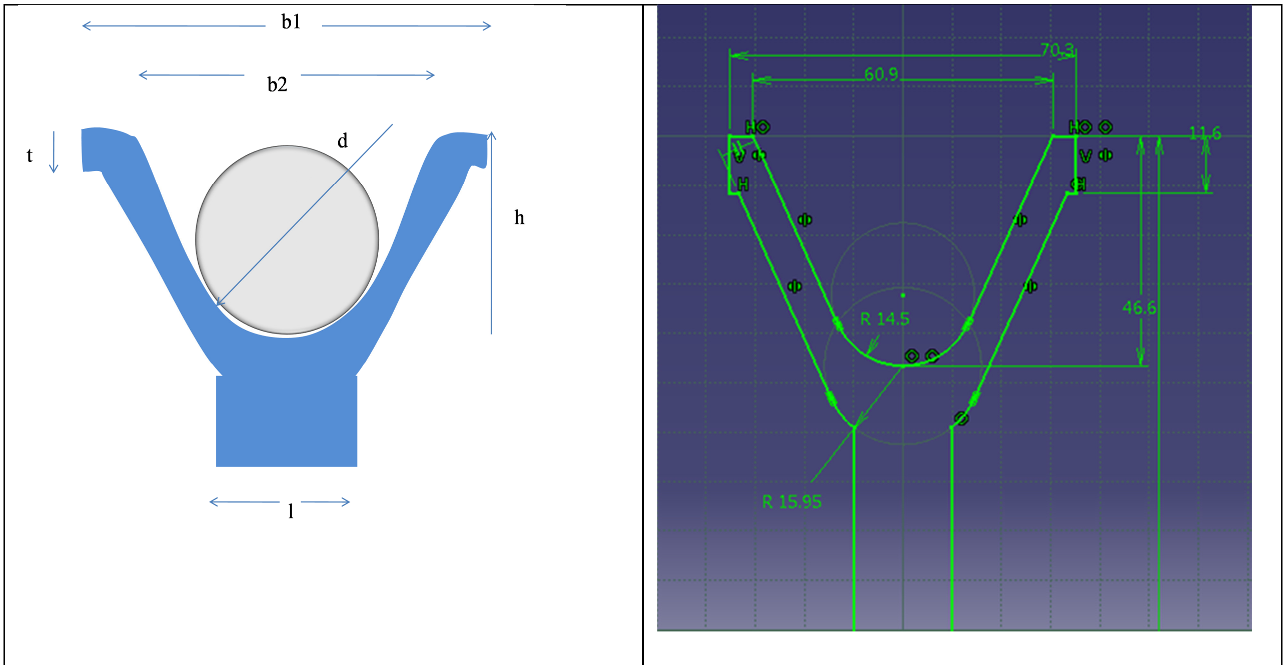


Fig 14: Pulley cross section

ROPE DRUM: The rope drum should be made of seamless pipe machined & grooved accurately, to -ensure proper seating of wire rope in a proper layer. The drum should be fitted with two heavy duty Ball / Roller bearings of reputed make for smooth operation & longer life.

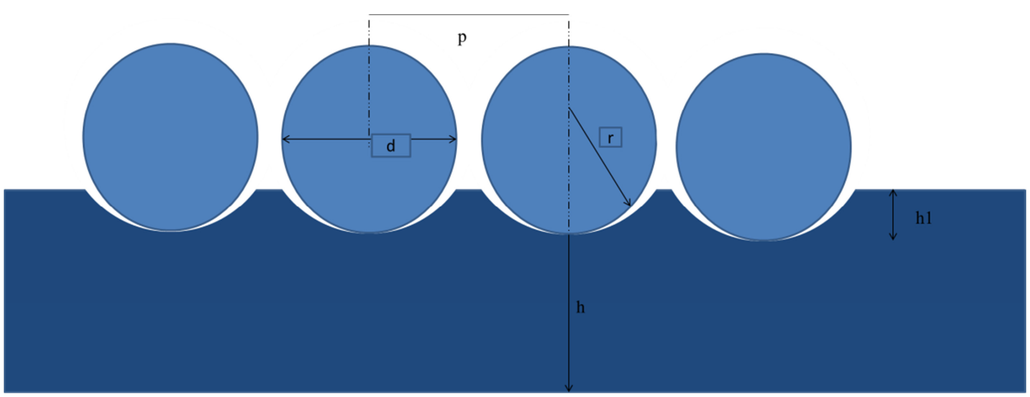


Fig 15: Rope drum cross section

symbol	Value in d	Value
Drum groove radius(r)	0.5d+1.5mm	16mm
Pitch(p)	2r+3mm	35mm
H1	0.25d	7.25
h	1.1d	31.9mm
Drum dia [d0]	23d to 30d Take 23d	667mm
Drum length		4197mm

Drum length =pitch x ground height x no of ropefall/drum dia =35x10000x8/667

=4197mm

[let ground height be 10m]

Average drum thickness = h+h/2 = 31.9+7.1/2 =35

Design of the MOTORS: Hoist & crane duty hour rated squirrel cage induction motors, confirming to IS 325 with comparatively higher H.P. and higher starting torque to reduce handling time. It is flange mounted to suit the design and provided with suitable insulation

Lifting speed varies from =10 t0 26 fpm

=50.79 to 132.08 mm per sec

Speed of drum =4x0.132/R=2πN

For drum rotating angular speed = w lifting speed/diameter of the drum =0.184 radian

Power transmitted by shaft =2πNT/60

Power = 4x0.132x50000x6 =158400Watt= 158.4 kW

Design of the Gear Box: Totally enclosed oil splash lubricated & dust free gear box should be provided for smooth, trouble free & longer life. All gears are helical type and cut from alloy steel / low carbon steel on hobbing machines for achieving higher precision & a special process of gear toughening ensures smooth, silent, trouble free running of drive system. The pinions and gears are supported on anti-friction bearings on both ends.

For drum rotating angular speed = wx lifting speed/diameter of the drum =0.184 radian/s

For motor let w_1=62.31radian/s

Reduction in speed =w_1/w =342 times

Hoist Brake

When selecting the proper brake for a specific application, there are several *factors are* consider; a few that need to be reviewed

- brake torque
- stopping time
- deceleration rates
- brake mounting
- brake location
- thermal rating
- nvironment
- brake style.

The brake systems manufactured external friction brakes.

Applications for which these brakes are suited can be classified into two general categories:

non-overhauling

overhauling.

A) Non-overhauling loads are typically horizontally moving masses such as crane bridges, crane trolleys,horizontal conveyors.

B) Overhauling loads tend to accelerate in speed if a brake is not present, examples of which are crane hoists, winches, lifts, and downhill conveyors.

Non-overhauling loads require brake torque only to stop the load and will remain at rest due to friction. Overhauling loads have two torque requirements; the first is braking torque required to stop the load, and the second is the torque required to hold the load at rest.

BRAKE TORQUE

The full-load torque of a motor is a function of horsepower and speed and is commonly used to determine a brake torque rating. The brake torque rating is to equal or exceed the full load torque of a motor. The formula to calculate the full load motor torque is as follows:

$$T_{break} = \frac{5250 \times HP \ \times S.F}{RPM}$$

where: 5250 = constant

HP = motor horsepower = power inWatt/ 746

RPM = speed of motor shaft

S.F. = application service factor(assumed to be 1.25)

T_{brake} = static brake torque

T_{brake}=5250x158400/746 x1.25/600 =232.225 Nm

CRANE HOIST BRAKING TORQUE

Sizing of crane hoist brakes is typically based upon full load hoisting torque. The following is a brief summary of guidelines for hoist brakes.

Each hoist on a crane should be equipped with at least one spring-set magnetic brake; hoists handling hot metal should be equipped with more than one brake. Brake rating expressed as a percent of hoisting torque at the point of brake application should be no less than the following:

1. 1.5 times when only one brake is used.
2. 1.5 times when multiple brakes are used and the hoist is not used to handle hot metal. Failure of any one brake should not reduce braking torque below normal
3. 1.75 times for hoists handling hot metal
4. Failure of any one brake shall not reduce brake torque below 1.5 times

Methods used a

Principle of bending of a beam with large initial curvature

Algebraic calculations

Empirical calculations

CATIA v 5 for 3D modeling,

ANSYS 12 for stress calculation and deformation analysis

LITERATURE SURVEY

RajendraParmanik in a post “Design Of Hoisting Arrangement Of E.O.T. Crane(2008) he has discussed about history of crane, various types of crane ,application and a model design of the various parts of the EOT crane

R. Uddanwadiker, in the paper "Stress Analysis of Crane Hook and Validation by Photo-Elasticity” states that “Crane Hooks are highly liable components and are always subjected to failure due to accumulation of large amount of stresses which can eventually lead to its failure By predicting the stress concentration area, the shape of the crane is modified to increase its working life and reduce the failure rates.”

Results

First the hook dimension was calculated and taken various cross section keeping bed diameter ,area of the cross section and depth constant and analyzed the stress

Result : stress in the rectangular section was minimum followed by trapezoidal section a bit more circular cross section was found to exert maximum stress

From the ANSYS analysis result was trapezoidal section was least stressed

But stress in the modified section was more because the cross sectional area was less in the modified section

Bed dia for hook =300mm

Depth=256mm

Wire rope diameter chosen 6x19 as it show less stress on the wires

Rope diameter=29mm

Increase in no of rope falls increase drum length ,work done ,

Drum length =4.5 m

Drum dia =0.676m

Pulley dia=0.725m

Motor power=158.4 KWatt

Braking torque=23 Nm

Conclusion

In the designed hoist model trapezoidal section show less stress

The modified section should show less stress but due to reduction in area it shows more stress

Using more no. of rope falls divide the load and make the tension less. Also it makes the work faster .E.g if we use 4 rope falls then using the same force 4 times work is done

But increase in rope fall increase the rope length by that times ,which is expensive

Also the rope length determine the drum length. Increase in drum length increase the volume of setup to reduce the volume we can double winding of rope on the drum can be adopted

Motor power required depends on lifting speed and load applied

The angular speed of drum and the motor are different so a gear box is used for power transmission

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